〔論　文〕

Improvement in Maneuverability and Stability of Vehicle through Front/ rear Active Steer Control with Steer-by-wire

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Abstract

Various techniques have been proposed for improving the maneuverability and stability of a vehicle through active steer control of front or rear wheels by using the steer-by-wire mechanism. We compare a performance feature of a front-wheel active steer, a rear-wheel active steer and a front/rear wheel active steer control, the merits and demerits, and evaluate the technical possibility. We take up some explored transfer functions of multiusing as a target performance of vehicle motion control. For instance, the target transfer function includes the first delay characteristic. And each control law of steering the front or rear wheels actively is derived to achieve such a target characteristic. Next, the differences of the influence that various active steer control systems exert on the vehicle motion characteristics are clarified by the theoretical calculation for the vehicle motion characteristics of yaw rate, lateral acceleration, and body slip angle etc., and steering effort characteristics.

Key Words：Automobile, Vehicle Dynamics, Motion Control, Four-Wheel-Control, Steer-By-Wire, Maneuverability, Stability

1. Introduction

It has been 20 years since the four-wheel steering that actively steered the rear wheels to improve the maneuverability and stability of automobiles was put to practical use[(1)-(4)]. Recently, advances have been made in the research and development of such mechanisms[(5)] and steer-by-wire [(6)-(8)] that can actively change the steering gear ratio for a steering mechanism of front wheels. Especially, the advancement of the hardware that focuses on the mechanism side is remarkable. Steering devices that actively control the front wheels, and four-wheel active steer system (4WAS) that actively controls both the front and rear wheels have been put to practical use, too. With the advancement of such hardware, a new steer control method has made the improvement of vehicle dynamics capability possible.

However, research comparing the influence on vehicle dynamics under the same control target performance for three methods of the front wheel active-steering[(9)-(11)], the rear wheel active-steering, and the front-rear wheel active-steering[(12)-(15)] has been limited when a variety of steer control methods are examined. Furthermore, there has been problems understanding each feature easily.

In this paper, we compare the performance features and the merits and demerits of the above-mentioned three steer controls under the same objective control performance, and discuss the

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possibility of improving maneuverability and stability using the active steer control. When the control law is derived, we take up the examples of setting the transfer function on yaw rate to steering wheel angle multiused as the target performance to the first delay characteristic etc. First, the front wheel active-steering, the rear wheel active-steering, and the front-rear wheel active-steering to achieve the target performance request each control law. Next, the vehicle dynamics characteristics such as yaw rate, lateral acceleration, body sideslip angle etc., and steering effort characteristics are compared by theoretical calculation. In addition, we clarify the differences of influence that various steering systems exert on vehicle performance. The simulation results show that the front-rear wheel active-steering control can greatly improve the maneuverability and stability of the vehicle.

2. Equations of motion

2.1 Nomenclature

The notations used by analysis model and the main parameters and characteristic values used to calculate are listed below.

C_i : sum of two cornering powers of right and left tires
{770*2, 1280*2 N/deg}
C_s : equivalent damping coefficient of steering wheel system
F_i : sum of two cornering forces of right and left tires
F_h : steering efforts $F_h = T_h/d_h$
G_{r0} : steady yaw rate gain of 2WS
I_h : moment of inertia around steering wheel shaft
I_z : yaw moment of inertia {2400kgm^2}
K_s : stability factor
N : steering overall gear ratio {15.4}
T_h : steering wheel steer torque
T_p : assistance output torque of power steering system
T_{pat} : output torque of pseudo reactive force device
a,b : distance from front/rear axle to center of gravity of vehicle {1.18, 1.44m}
d_h : diameter of steering wheel {0.28m}
l : wheelbase {2.62m}
m : vehicle mass {1500kg}
r : yaw rate
s,t : Laplace transform operator, time
t_c : caster trail of front wheel {0.03m}
t_{pf} : pneumatic trail of front tire {0.03m}
v : vehicle velocity
α_y : lateral acceleration of vehicle
β : body sideslip angle at c.g. of vehicle
β_i : tire sideslip angle
δ_i : front/rear wheel steer angle
θ : steering wheel angle
τ_r : time constant of yaw rate time-lag of first order characteristic

Subscript

i : f-front wheels, r-rear wheels

Coordinates

o-xy: coordinates fixed to the vehicle body

2.2 Vehicle

The calculating model for the analysis of vehicle dynamics is illustrated in Figure 1. When the cornering characteristics of right and left tires are assumed to be the same, the motion equations are as follows.

$$mv(\dot{\beta}+r)=F_f+F_r \tag{1}$$

$$I_z\dot{r}=aF_f-bF_r \tag{2}$$

Each cornering force generated in the front and rear tires is given by the following equation.

$$\left.\begin{array}{l} F_f=C_f\beta_f \\ F_r=C_r\beta_r \end{array}\right\} \tag{3}$$

Eq.(4) yields the sideslip angle of each tire in Eq. (3).

$$\left.\begin{array}{l} \beta_f=\delta_f-\beta-\dfrac{a}{v}r \\ \beta_r=\delta_r-\beta+\dfrac{b}{v}r \end{array}\right\} \tag{4}$$

Eqs.(1)-(4) are combined in the form of a matrix equation on the body sideslip angle and the yaw rate, and the following equation is obtained[16].

$$\begin{bmatrix}\beta \\ r\end{bmatrix}=\begin{bmatrix}p_{11}(s) & p_{12}(s) \\ p_{21}(s) & p_{22}(s)\end{bmatrix}\begin{bmatrix}\delta_f \\ \delta_r\end{bmatrix}=P(s)\begin{bmatrix}\delta_f \\ \delta_r\end{bmatrix} \tag{5}$$

where

$$\left.\begin{array}{l} p_{11}(s)=\dfrac{G_{\beta1}\omega_n^2(\tau_{\beta1}s+1)}{s^2+2\zeta\omega_ns+\omega_n^2} \\ p_{12}(s)=\dfrac{G_{\beta2}\omega_n^2(\tau_{\beta2}s+1)}{s^2+2\zeta\omega_ns+\omega_n^2} \\ p_{21}(s)=\dfrac{G_{r1}\omega_n^2(\tau_{r1}s+1)}{s^2+2\zeta\omega_ns+\omega_n^2} \\ p_{22}(s)=\dfrac{G_{r2}\omega_n^2(\tau_{r2}s+1)}{s^2+2\zeta\omega_ns+\omega_n^2} \end{array}\right\} \tag{6}$$

The relations between the state variables β, r and the steer angles δ_f, δ_r can be understood by Eq. (5). The block diagram showing the relations between these is illustrated in Figure 2. The relational expressions of notations in Eqs.(5) and (6) are as follows.

$$G_{\beta 1}=\frac{1-\dfrac{m}{l}\cdot\dfrac{av^2}{bC_r}}{1+K_s v^2}\cdot\frac{b}{l}\qquad \tau_{\beta 1}=\frac{I_z v}{lbC_r-mav^2}\qquad G_{\beta 2}=\frac{1+\dfrac{m}{l}\cdot\dfrac{bv^2}{aC_f}}{1+K_s v^2}\cdot\frac{a}{l}\qquad \tau_{\beta 2}=\frac{I_z v}{laC_f+mbv^2}$$

$$G_{r1}=\frac{1}{1+K_s v^2}\cdot\frac{v}{l}\qquad \tau_{r1}=\frac{mav}{lC_r}\qquad G_{r2}=\frac{-1}{1+K_s v^2}\cdot\frac{v}{l}\qquad \tau_{r2}=\frac{mbv}{lC_f}$$

$$K_s=\frac{m}{l^2}\left(\frac{b}{C_f}-\frac{a}{C_r}\right)\qquad \omega_n=\sqrt{\frac{C_f C_r l^2}{mI_z v^2}-\frac{aC_f-bC_r}{I_z}}$$

$$\varsigma=\frac{\left(ma^2+I_z\right)C_f+\left(mb^2+I_z\right)C_r}{2l\sqrt{mI_z C_f C_r\left(1+K_s v^2\right)}}\qquad \zeta\omega_n=\frac{m\left(a^2C_f+b^2C_r\right)+I_z\left(C_f+C_r\right)}{2mI_z v}$$

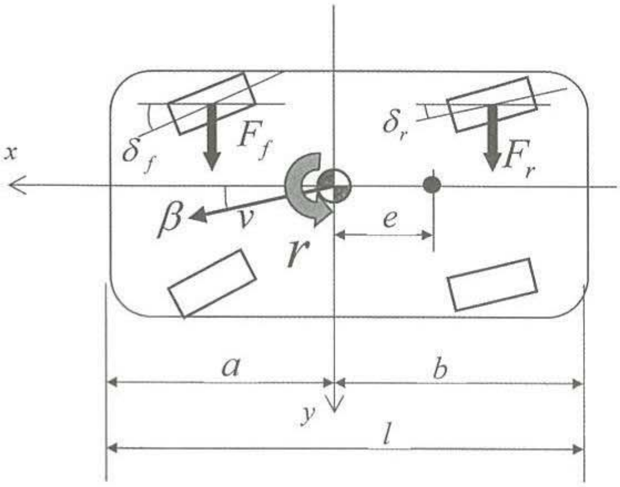

Fig.1 Theoretical analysis model

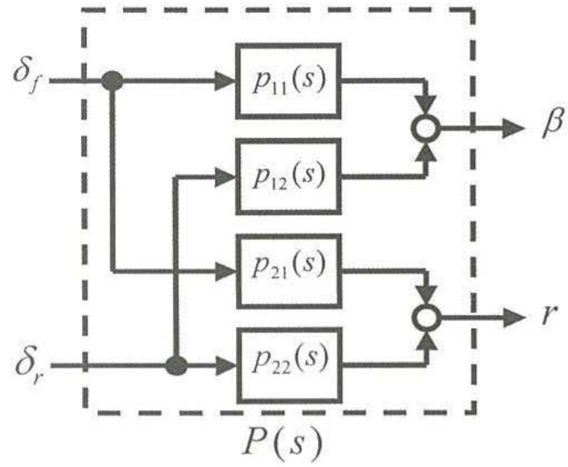

Fig.2 Block diagram of vehicle dynamics

2.3 Steering system

The transfer functions between steering wheel angle θ and front/rear wheel steer angle δ_f, δ_r are defined respectively as $G_f(s)$ and $G_r(s)$.

$$\left.\begin{aligned}\delta_f&=G_f(s)\theta\\ \delta_r&=G_r(s)\theta\end{aligned}\right\}\tag{7}$$

Figure 3 shows a block diagram that added the relation of Eq.(7) to Figure 2. Because a conventional vehicle is 2WS (two-wheel-steering) whose front wheels are a steering wheel system of pure machine type and the rear wheels are not steered, the transfer functions are given by $G_f(s)=1/N$, $G_r(s)=0$. Moreover, when the front wheel or the rear wheel steering is actively controlled by the SBW(steer-by-wire) mechanism. $G_f(s)$ and $G_r(s)$ show the control functions.

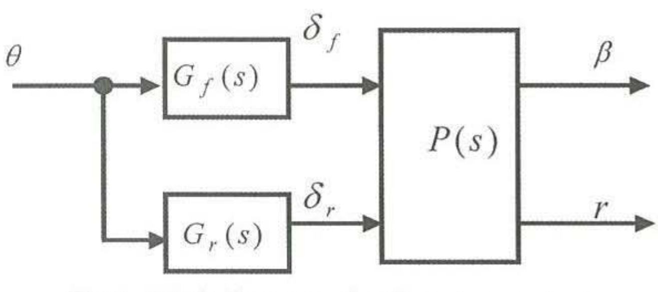

Fig.3 Block diagram of active steer control

Next, we think about an easy model to analyze the characteristics of the steering effort and power to maintain the rotation of steering wheel when a driver maneuvers the steering wheel. The analysis model of a general pure machine type steering system is illustrated in Figure 4. Considering assistance torque T_p of power steering system, the following equation yields the motion expression of this steering system.

$$I_h\ddot{\theta}+C_s\dot{\theta}+\left(t_c+t_{pf}\right)F_f/N=T_h+T_p \tag{8}$$

We also consider about the steer-by-wire system shown in Figure 5. SBW mechanically separates a part of rotating the steering wheel and a part of steering the front tires by the electric motor only according to the control signals on the rotation of the steering wheel. In this case, the driver can't obtain so-called road information because the force generated between the road and the tire is not mechanically fed back to the steering effort and adequate operation becomes difficult.

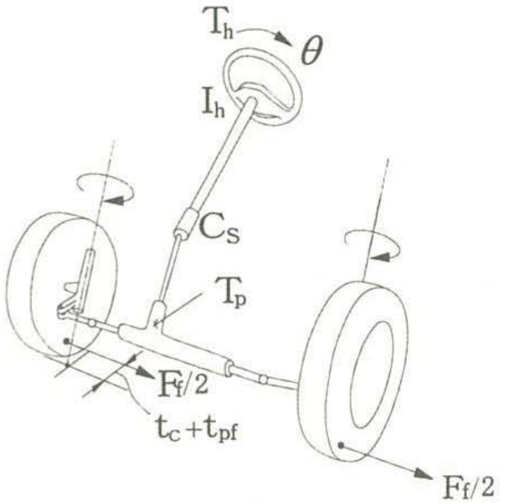

Fig.4 Conventional steering system model

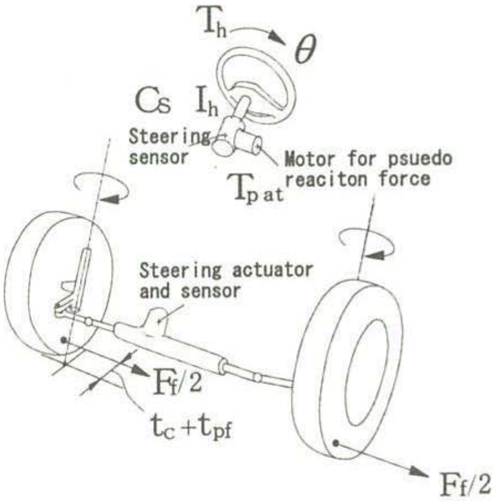

Fig.5 Steer-by-wire system model

Then, a pseudo reactive force device that obtains appropriate steering effort by giving a force corresponding to tire generation forces, vehicle velocity, steering wheel angle etc. is necessary for SBW. Applying the device that generates a pseudo reactive torque T_{pat} according to the tire generation forces, the vehicle velocity and the steering wheel angle, we obtain the following equation of SBW.

$$I_h\ddot{\theta}+C_s\dot{\theta}+T_{pat}\left(F_f,v,\theta\right)=T_h \tag{9}$$

The pseudo reactive force device is necessary to give a pseudo reactive torque characteristic T_{pat} $\left(F_r,v,\theta\right)$ to obtain the steering wheel maneuver characteristic similar to a usual pure machine type steering system with the power steering device so that the sense of incompatibility should not occur in the steering wheel operation.

3. Steer control law

We determine the control law of various steer controls of the first delay characteristic [17] of yaw rate and yaw center position control [18]-[20] which is often used as the target characteristics of vehicle dynamics control. In addition, body sideslip angle at the center of gravity, yaw rate and lateral acceleration are derived. To facilitate the comparison of analytical results, the control laws are requested as a feed forward method on steering wheel angle shown by Eq.(7).

3.1 Control to be the first delay characteristic of yaw rate

The steer control laws are led to set a first delay characteristic where yaw rate to steering wheel angle is shown by Eq.(10).

$$\hat{r}_m(s) = \frac{r_m(s)}{\theta(s)} = \frac{G_{r0}}{1+\tau_r s} \tag{10}$$

Or, the target characteristic is given by the next equation.

$$r_m(s) = \hat{r}_m(s)\theta \tag{11}$$

where G_{r0} is a steady gain of yaw rate characteristic on 2WS, and it is given by the following equation. Moreover, τ_r is a first delay time constant of the yaw rate characteristic.

$$G_{r0} = \frac{v}{Nl\left(1+K_s v^2\right)}$$

① Front wheel active-steering

Substituting the relations of $\delta_f = G_f(s)\theta$, $\delta_r = 0$ and Eq.(10) into Eq.(5), we obtain the following control function of the front wheel active-steering.

$$G_f(s) = \frac{\hat{r}_m(s)}{p_{21}(s)} \tag{12}$$

Here, two transfer functions of the body sideslip angle and the lateral acceleration to steering wheel angle are given respectively by the next equation.

$$\left.\begin{aligned} \frac{\beta(s)}{\theta(s)} &= \frac{p_{11}(s)\hat{r}_m(s)}{p_{21}(s)} \\ \frac{\alpha_y(s)}{\theta(s)} &= v\left(s\frac{\beta}{\theta}+\frac{r}{\theta}\right) = \frac{v\left(sp_{11}(s)+p_{21}(s)\right)\hat{r}_m(s)}{p_{21}(s)} \end{aligned}\right\} \tag{13}$$

The block diagram of the front wheel active steer control is illustrated in Figure 6.

② Rear wheel active-steering

Substituting the relations of $\delta_f = \frac{1}{N}\theta$, $\delta_r = G_f(s)\theta$ and Eq.(10) into Eq.(5), we obtain the following control function of the rear wheel active-steering.

$$G_r(s) = \frac{N\hat{r}_m(s) - p_{21}(s)}{Np_{22}(s)} \tag{14}$$

Here, two transfer functions of the body sideslip angle and the lateral acceleration to steering wheel angle are given respectively by the next equation.

$$\left.\begin{aligned}\frac{\beta(s)}{\theta(s)}&=\frac{p_{11}(s)p_{22}(s)-p_{12}(s)p_{21}(s)+Np_{12}(s)\hat{r}_m(s)}{Np_{22}(s)}\\ \frac{\alpha_y(s)}{\theta(s)}&=v\left\{\frac{s(p_{11}(s)p_{22}(s)-p_{12}(s)p_{21}(s)+Np_{12}(s)\hat{r}_m(s))}{Np_{22}(s)}+\hat{r}_m(s)\right\}\end{aligned}\right\} \tag{15}$$

The block diagram of the rear wheel active steer control is illustrated in Figure 7.

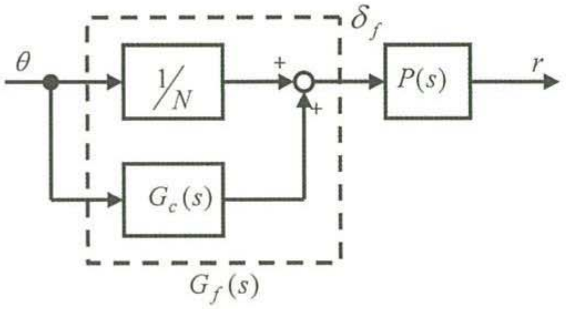

Fig.6 Block diagram of front wheel active-steering

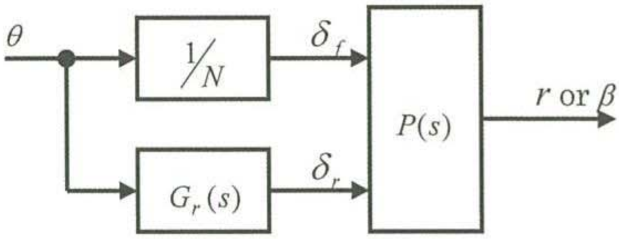

Fig.7 Block diagram of rear wheel active-steering

③ Front-rear wheel active-steering

Because two target characteristics can be obtained for a front-rear wheel active steer, the target characteristics of yaw rate and body sideslip angle are made here. The body slip angle characteristic β_m that aims is shown by $\beta_m(s)=\hat{\beta}_m(s)\theta$. Substituting Eq.(7) into Eq.(5), the following equation is formed.

$$\begin{bmatrix}\hat{\beta}_m(s)\\ \hat{r}_m(s)\end{bmatrix}=P(s)\begin{bmatrix}G_f(s)\\ G_r(s)\end{bmatrix} \tag{16}$$

Transforming Eq.(16) with respect to the control functions of the front and rear wheels, we obtain Eq.(17).

$$\begin{bmatrix}G_f(s)\\ G_r(s)\end{bmatrix}=P^{-1}(s)\begin{bmatrix}\hat{\beta}_m(s)\\ \hat{r}_m(s)\end{bmatrix}=\frac{1}{|P(s)|}\begin{bmatrix}p_{22}(s) & -p_{12}(s)\\ -p_{21}(s) & p_{11}(s)\end{bmatrix}\begin{bmatrix}\hat{\beta}_m(s)\\ \hat{r}_m(s)\end{bmatrix} \tag{17}$$

Here, the transfer function of lateral acceleration is given by the next equation.

$$\frac{\alpha_y(s)}{\theta(s)}=v\left(s\frac{\beta_m}{\theta}+\frac{r_m}{\theta}\right)=v\left(s\hat{\beta}_m(s)+\hat{r}_m(s)\right) \tag{18}$$

3.2 Control which makes yaw center a target position

A yaw center is at the position of the distance of ***e*** from the body center of gravity to the rear side, $\dot{y}-er=0$ is approved. And the following relation is obtained.

$$\beta=er/v \tag{19}$$

We obtain the steer control laws to fix a yaw center position to an arbitrary position by using this relational equation. Especially, the control of $\boldsymbol{e}=0$ is the same as that which makes the body slip angle at the vehicle center of gravity zero known best.

① Front wheel active-steering

Substituting $\delta_f = G_f(s)\theta, \delta_r = 0$ into Eq.(5), the relations of $\beta = p_{11}\delta_f$, $r = p_{12}\delta_f$ are obtained. $p_{11}\delta_f v = p_{12}\delta_f e$ is given by Eq.(19), and $(p_{11}v - p_{12}e)\delta_r = 0$ consists.

Here, because δ_f is not zero, $p_{11}v - p_{12}e = 0$ always consists. Therefore, $e = p_{11}v/p_{12}$ is led. However, because e is constant, this is contradicted. In a word, the solution does not exist.

It becomes $e = p_{11}v/p_{12}$ even if it controls to obtain the yaw rate characteristic of the following Eq.(21), and this result is the same. Control that makes yaw center a target position cannot be theoretically done through the front wheel active steer control.

② Rear wheel active-steering

Substituting the relations of $\delta_f = \frac{1}{N}\theta$, $\delta_r = G_r(s)\theta$ and Eq.(19) into Eq.(5), we obtain the following control function of the rear wheel active-steering.

$$G_r(s) = \frac{vp_{11}(s) - ep_{21}(s)}{N(ep_{22}(s) - vp_{12}(s))} \tag{20}$$

The main state variables of vehicle motion are shown by the next transform functions.

$$\left.\begin{aligned}
\frac{r(s)}{\theta(s)} &= \frac{v(p_{11}(s)p_{22}(s) - p_{12}(s)p_{21}(s))}{N(ep_{22}(s) - vp_{12}(s))} = \frac{v|P(s)|}{N(ep_{22}(s) - vp_{12}(s))} \\
\frac{\beta(s)}{\theta(s)} &= \frac{e}{v}\cdot\frac{r}{\theta} = \frac{e|P(s)|}{N(ep_{22}(s) - vp_{12}(s))} \\
\frac{\alpha_y(s)}{\theta(s)} &= (se+v)\frac{r}{\theta} = \frac{(se+v)v|P(s)|}{N(ep_{22}(s) - vp_{12}(s))}
\end{aligned}\right\} \tag{21}$$

Developing Eq.(20) and Eq.(21) yields Eq.(22) and Eq.(23) respectively.

$$G_r(s) = \frac{C_f\{(mae - I_z)s + mav + lC_r(e-b)/v\}}{NC_r\{(mbe + I_z)s + mbv + lC_f(e+a)/v\}} \tag{22}$$

$$\left.\begin{aligned}
\frac{r(s)}{\theta(s)} &= \frac{C_f l}{N\{(mbe + I_z)s + mbv + lC_f(e+a)/v\}} \\
\frac{\beta(s)}{\theta(s)} &= \frac{eC_f l}{vN\{(mbe + I_z)s + mbv + lC_f(e+a)/v\}} \\
\frac{\alpha_y(s)}{\theta(s)} &= \frac{C_f l(se+v)}{N\{(mbe + I_z)s + mbv + lC_f(e+a)/v\}}
\end{aligned}\right\} \tag{23}$$

③ Front-rear wheel active-steering

Substituting $\beta_m = er_m / v$ sought from Eq.(19) into Eq.(17), we can obtain the control function of the following front-rear active-steering.

$$\left.\begin{aligned} G_f(s) &= \frac{\hat{r}_m(s)(ep_{22}(s) - vp_{12}(s))}{v|P(s)|} \\ &= \frac{G_{r0}\{(mbe + I_z)s + mbv + lC_f(e+a)/v\}}{lC_f(1+\tau_r s)} \\ G_r(s) &= \frac{\hat{r}_m(s)(vp_{11}(s) - ep_{21}(s))}{v|P(s)|} \\ &= \frac{G_{r0}\{(mae - I_z)s + mav + lC_r(e-b)/v\}}{lC_r(1+\tau_r s)} \end{aligned}\right\} \tag{24}$$

Here, the transfer functions of the body sideslip angle and the lateral acceleration to steering wheel angle are given respectively by the next equation.

$$\left.\begin{aligned} \frac{\beta(s)}{\theta(s)} &= \frac{e\hat{r}_m(s)}{v} \\ \frac{\alpha_y(s)}{\theta(s)} &= (se + v)\hat{r}_m(s) \end{aligned}\right\} \tag{25}$$

4. Calculation and Consideration

Maneuverability and stability of vehicles through the front wheel active-steering (called FAS in this paper), the rear wheel active-steering (called RAS) and the front-rear active-steering (called FRAS) are compared with those of 2WS by computer simulation. To compare three kinds of steer controls, whose objective characteristic is set to the first delay characteristics of yaw rate shown by Eq.(10). In addition, The control law of FRAS uses the proposed control functions that are determined so that the yaw center can coincide with the vehicle's center of gravity at all times. In a word, we set up that the yaw center position is $e = 0$. In this calculation, G_{r0} of Eq.(10) is set in a steady yaw rate gain of 2WS and the time constant of first-order delay τ_r is set to 0.05 second.

First of all, the step response and the frequency response to steering wheel angle are examined at vehicle velocity 120 km/h. Next, we evaluate the driver-vehicle system that considers driver's characteristic by computer simulation of a lane change.

4.1 Control response and stability

Figure 8 shows the step response characteristics for the steering wheel angle 30 deg. Three kinds of steer controls perform that the yaw rate follows a target characteristic. FRAS can keep the body sideslip angle to zero at all times, and this means that the yaw center is always at the vehicle's center of gravity. The transient states are obviously different though the static characteristics of vehicle motion state variables become the same in three kinds of steer controls. Standing up of the lateral acceleration is fast in the order of FRAS, RAS, 2WS, and FAS. The transient overshoot of the yaw rate and the lateral acceleration disappears when the steer controls are done. The body slip angle of RAS is smaller than that of FAS in the transient state.

Figure 9 shows the frequency response characteristics of vehicle motion state variables. The unit of the gain of yaw rate and lateral acceleration characteristics is [1/s] and [G/deg] respectively. As for FRAS, both decreases in the gain of the yaw rate and the lateral acceleration are small, and the phase lags are also small even if the steer frequency rises. The yaw rate characteristics of FAS and RAS show the same characteristic as FRAS according to the target. However, the lateral acceleration characteristics differ greatly, and RAS have excellent characteristics as the decrease in the gain is smaller than FAS, and the phase lag is also smaller, too. And, we can understand that FAS has the lateral acceleration characteristics with the tendency to look like 2WS.

Evaluating the steer control response and stability, we obtained the result as FRAS, RAS, FAS, and 2WS in order.

4.2 Steering maneuver characteristics

Figure 10 shows the step response characteristics of front and rear wheel steer angles. The appearance from which the wheels are steered to the opposite phase momentarily assisting can be understood at all steer controls. Especially, FAS needs the biggest real steer angle of front wheels compared with other steer controls operation start of the steering wheel. As for FRAS, when it takes the average total time, both front and rear wheels are steered by the biggest steer angles. The control steer angles of front and rear wheels are always more necessary for the steady cornering. On the other hand, the control steer angles of FAS and RAS are zero during the steady cornering.

Figure 11 shows the frequency response characteristics of the front and rear wheel steer angles. As for FAS and RAS, when the steer frequency rises, the gain grows. The phase advances in FAS in the area where the steer frequency is high, and RAS has a characteristic advanced by the phase in the low area. FRAS has the largest gain on both the front wheels and rear wheels.

FAS needs the big control steer angles in a area where the steer frequency is high though the size of tire cornering force, the steer control law are related. FRAS always needs the big control steer angles in both the front wheels and rear wheels, and becomes the most disadvantageous steer control technique if we consider the energy consumption of actuator, capacity, and the packaging on the body. Thinking about practical use, it is indispensable to consider the review of the control law etc..

In addition, enhancing the meaning of steering gear ratio for the pure machine type steering device, we define the value in which the steering wheel angle is divided in a front wheel real steer angle as an equivalent steering gear ratio. The reciprocal of the gain in Figures 8 and 9 is the characteristic of the equivalent steering gear ratio. The shape of waves of the step response provides evidence that a device which can do the steering gear ratio in changeability at time is necessary to achieve FRAS and FAS.

Next, Figures 12 and 13 show the step response and the frequency response characteristics of steering effort respectively. Eq.(8) was used for the calculation. Because the decrease in the gain of the steer frequency 1Hz neighborhood is the smallest, FRAS may be the smallest decrease in working of felt steering from the steering effort change. RAS has some characteristics of a better tendency than FAS.

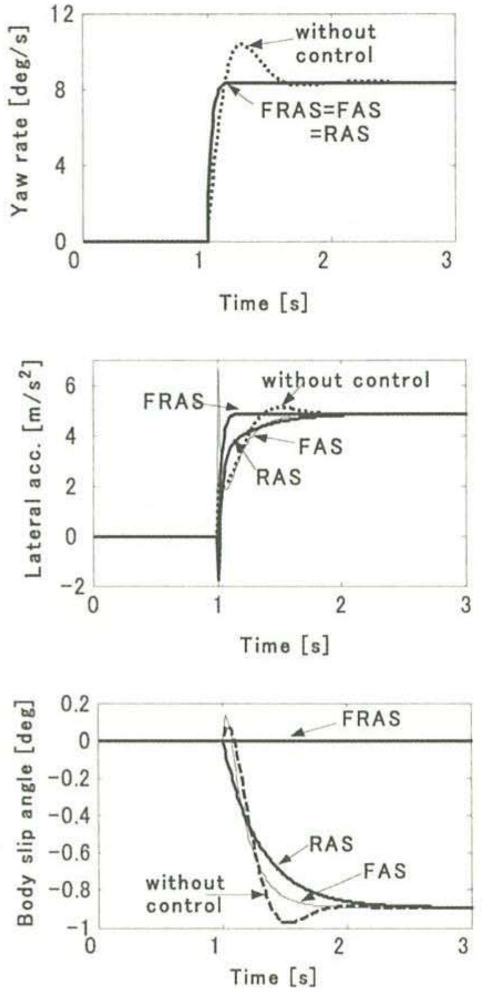

Fig.8 Step response characteristics

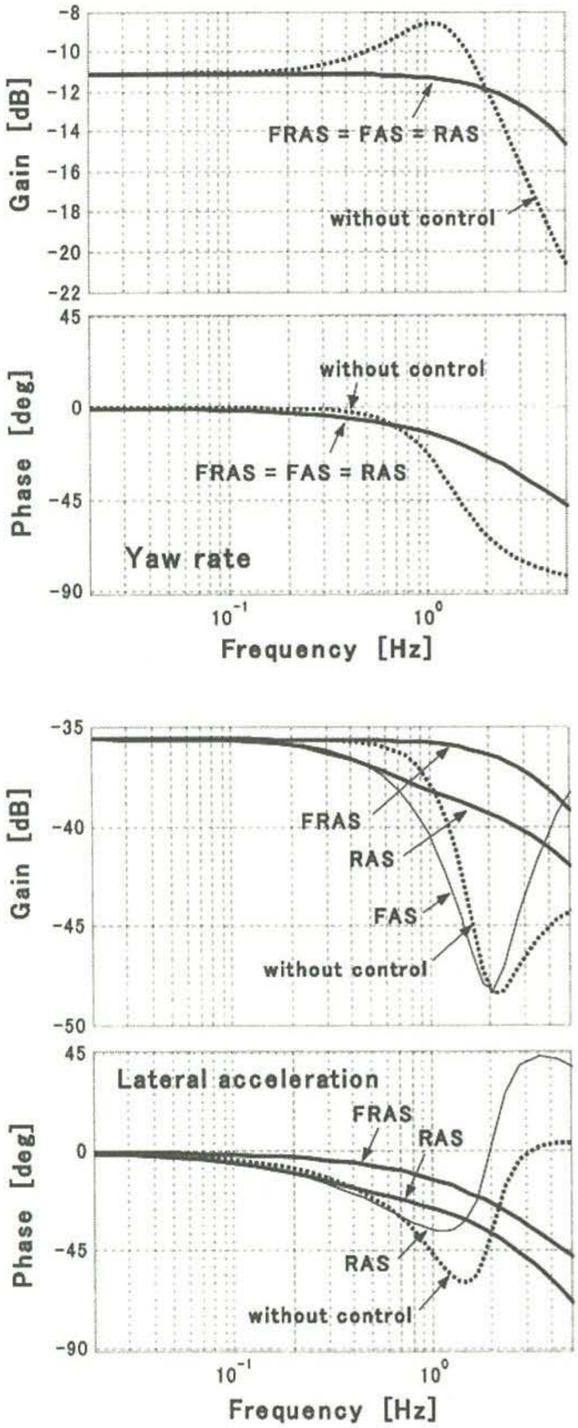

Fig.9 Frequency response characteristics

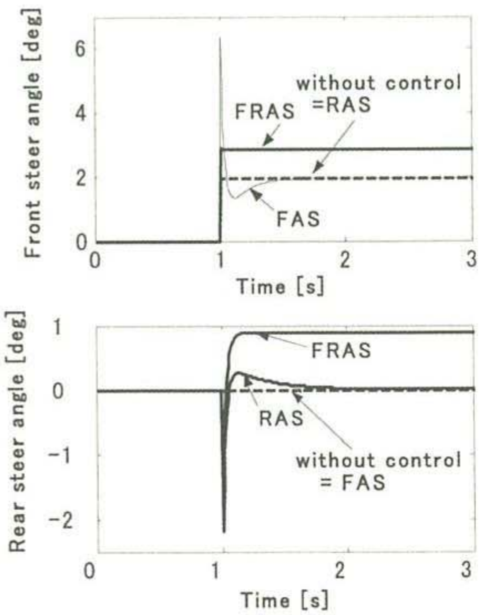

Fig.10 Step response characteristics of front and rear steering angles

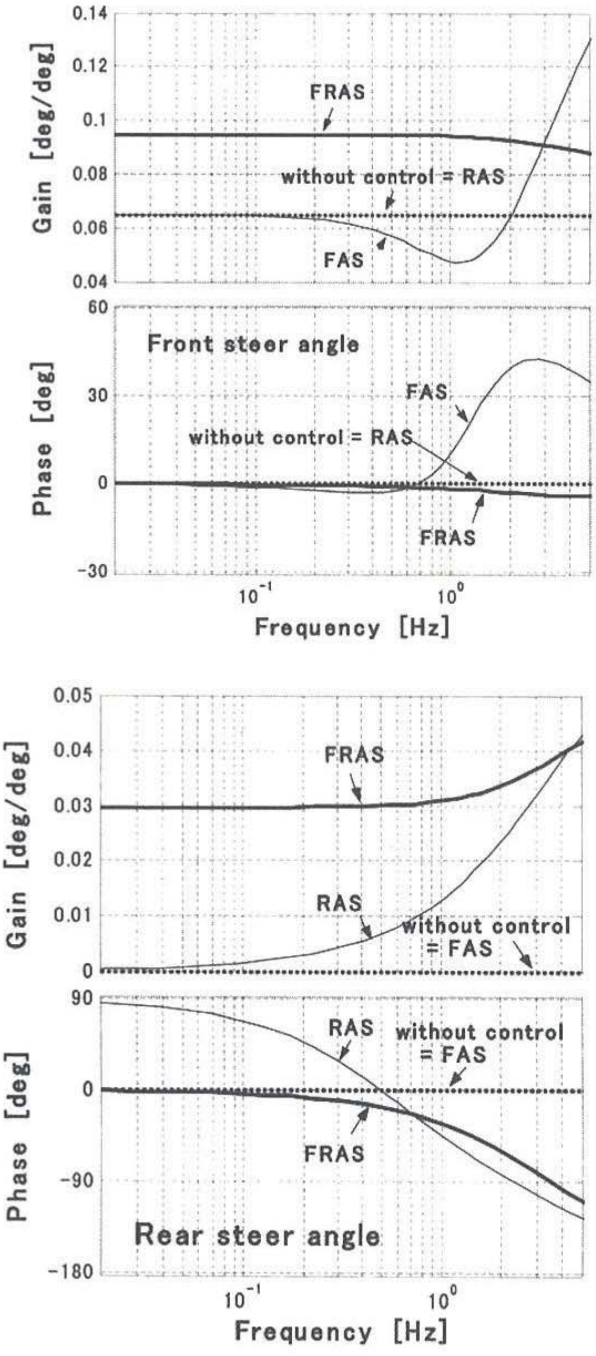

Fig.11 Frequency response characteristics of front and rear steering angles

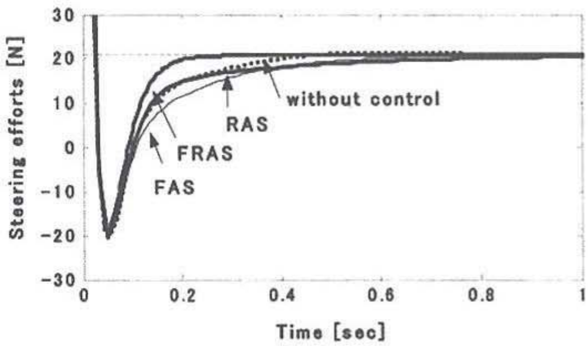

Fig.12 Step response characteristics of steering efforts

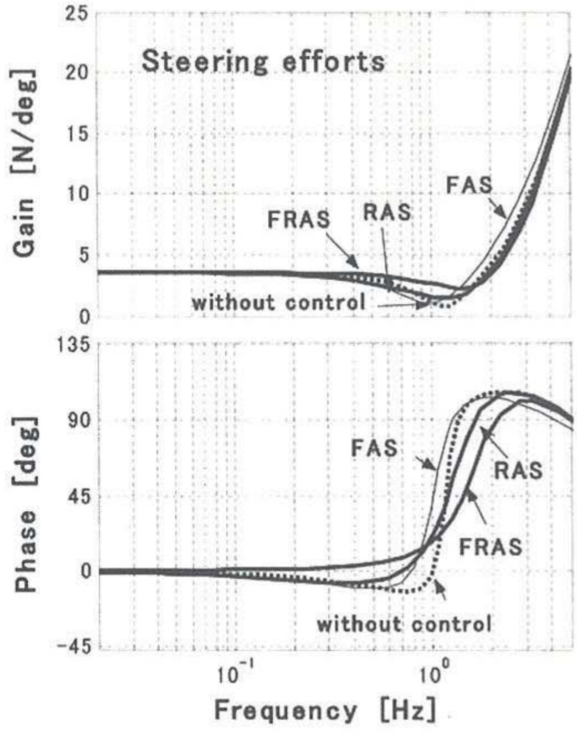

Fig.13 Frequency response characteristics of steering efforts

4.3 Closed loop characteristics

To investigate the control performance of the driver-vehicle system as a closed loop control system, we carry out computational simulation using the Runge-Kutta method in which the driver is asked to execute lane changes during quick braking. This running situation of vehicle simulates the maneuverability and stability during emergency obstacle avoidance.

The lane changing with braking by deceleration 0.3G at an initial velocity of 120 km/h is simulated. We apply the first-order prediction model using the feedback of lateral error from a desired course to the driver's fixation point as driver's control action model.

The driving lane where test vehicles run is illustrated in Figure 14. Figure 15 and 16 show the calculated result of the vehicular swept path of the vehicle's center of gravity while lane changing and the vehicle's dynamic characteristics respectively.

As for FRAS, the amount of maneuvering to the steering wheel can be a little, and a steady operation without impossibility can be done. Moreover, the standing up of the yaw rate, the lateral acceleration and the body sideslip angle to steering input is early, and the absolute values of these characteristics are small. We can understand that FRAS has few uselessness in the movement of the vehicle.

The evaluation results of the closed loop test is FRAS, RAS, FAS, and 2WS in order. But, it has been clarified that FRAS needs the biggest steer angle of front wheels in one side. In addition, FRAS needs a larger rear wheel control angle compared with RAS.

5. Conclusions

In the case of setting the transfer function on yaw rate characteristics to steering wheel angle to the first delay characteristics and the yaw center to a position as the target performance, we have requested each control law of the front wheel active-steering, the rear wheel active-steering and the front-rear wheel active-steering. And we have clarified the following by the process of obtaining theoretical expressions on the vehicle motion characteristics of yaw rate, lateral acceleration and body sideslip angle etc..

(1) When the target of yaw rate is the first delay characteristic, we can obtain the control law of the rear wheel active-steering and the front-rear wheel active-steering respectively.

(2) However, we cannot seek the control law of the front wheel active-steering for the yaw center position control method including the control that set the body sideslip angle at the center of gravity of body zero theoretically. In a word, the front active-steering cannot achieve a yaw center position control. The rear wheel active-steering and the front-rear wheel active-steering are feasible.

In addition, when the above-mentioned first delay characteristic of yaw rate was set to a target, computer simulation clarified the difference of the influence that various steer control systems exerted on the vehicle motion performance. The following results have been obtained. Here, both the first delay characteristic of yaw rate and yaw center $e = 0$ were set to an objective performance in the front-rear wheel active steer control.

(3) Comparing the stated variables of vehicle motion by three kinds of steer controls, the steady characteristics are the same. However, the transition characteristics and the frequency characteristics of lateral acceleration and body sideslip angle are greatly different. The evaluation results on the maneuverability and stability is the front-rear wheel active-steering, the rear wheel

active-steering, the front wheel active-steering and 2WS in order. Especially, the vehicle applied with front-rear wheel active-steering provides the great improvement of motion performance.

(4) As for the front wheel active-steering, the biggest steer angle of the front wheels is more necessary than that of the other steer control systems for starting the operation of the steering wheel. Investigating the average time on the amount of front and rear wheel steer angles, the front-rear wheel active-steering needs the biggest control angle in both front wheel and rear wheel, and the control angle of front and rear wheels is always necessary for steady cornering. The control angles of both the front active steer control and the rear active steer control become zero during steady cornering.

(5) Similarly in the test of closed loop system, the biggest control angle of the front-rear wheel active-steering is more necessary than that of the other steer control systems in both front wheel and rear wheel. On the other hand, as for the vehicle motion performance, the front-rear wheel active-steering has the best result. Next, it is the rear wheel active-steering, the front wheel active-steering and 2WS in order.

In this research, we executed the theoretical analysis by assuming the tire cornering characteristics to be linear and using the two-wheel model of two degree of freedom for vehicle dynamics to facilitate the comparison of various steer controls. The following problems include the maneuverability and stability in an area that the tire cornering characteristics are nonlinear or near the limit, the analysis of cornering performance with braking/driving etc.. Moreover, the front active steer control system needs to examine the operational feeling of the steering wheel and the pseudo reactive force characteristics of SBW. In addition, the front-rear wheel active-steering needs to examine a control law that can coexist with the improvement of vehicle dynamic performance and the small control steer angles.

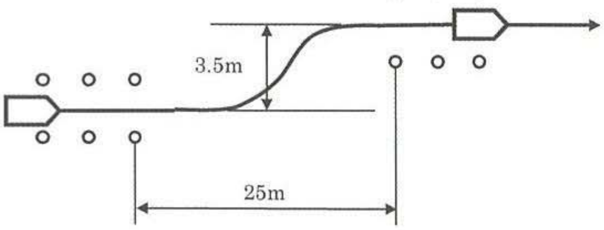

Fig.14 Lane changing test course

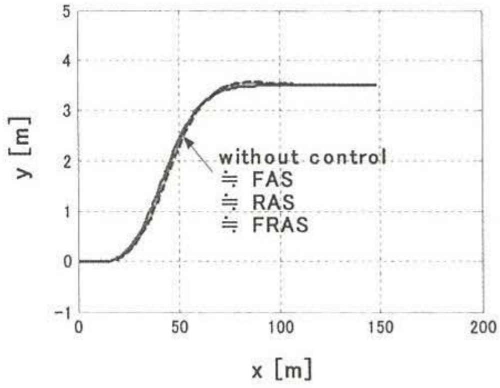

Fig.15 Track of lane changing

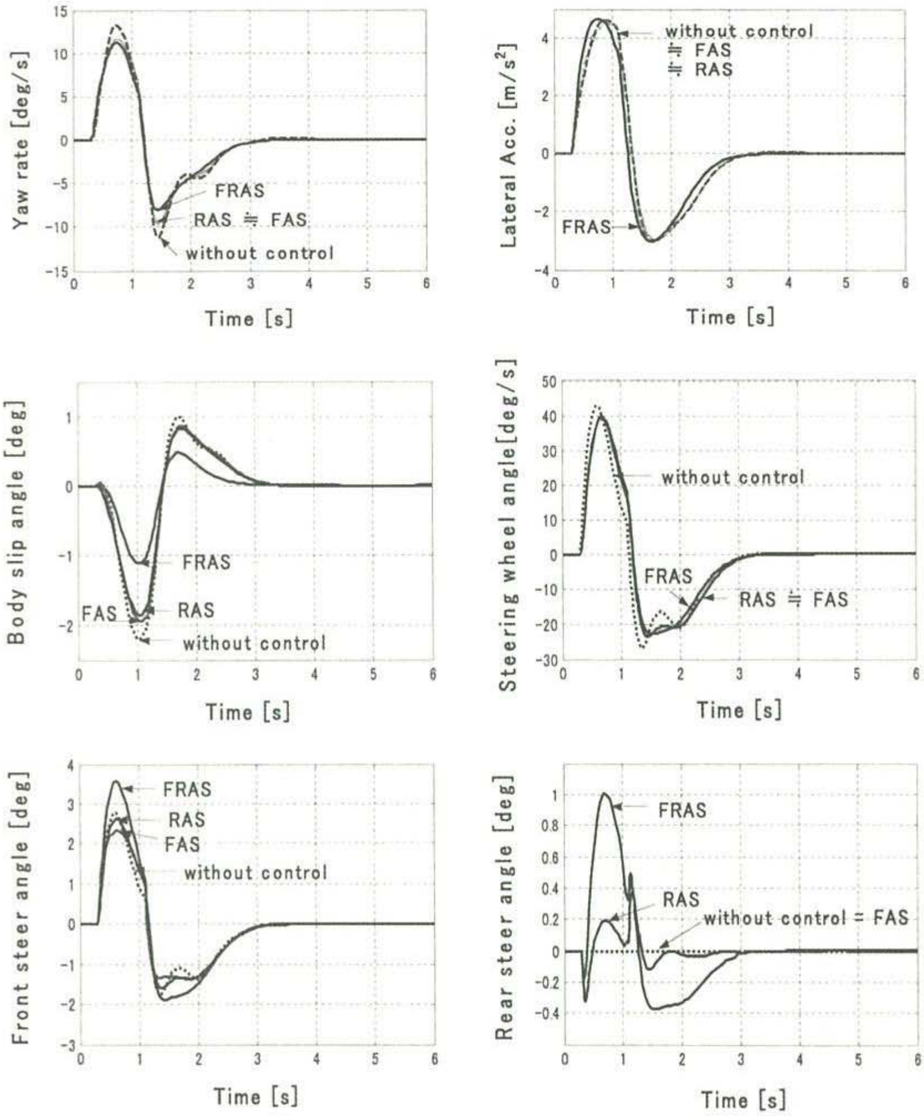

Fig.16 Simulation results of lane changing

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